CONDITION MONITORING OF ROLLING ELEMENT BEARINGS IN A GEARED ROTOR

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CERTIFICATE

This is to certify that the work presented in this thesis entitled "Condition Monitoring Of Rolling Elenent Bearings In A Geared Rotor" by LT RUPENDRA KAUSHIK has been carried out under my supervision and it has not been submitted elsewhere for the award of a degree.

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Rupendra Kaushik

CONTENTS

ABSTRACT

This study concerns itself with vibration based fault diagnosis of rolling element bearings in a geared rotor. A test rig is fabricated and installed. Instrumentation is set up. Investigations are carried out for three kinds of defects in ball bearings and vibration spectra are obtained at various rotor speeds for comparison with those from healthy bearing. Analysis is carried out in low, medium and high frequency ranges. Characteristic frequencies are identified and monitored for faults. An attempt has been made to associate the severity of bearing faults with vibration characteristics of the spectra in the three regimes frequency analysis.

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CHAPTER 1

INTRODUCTION

Vibration analysis has proved to be a powerful tool for analysis of failure in rotating machinery. Vibration analysis with appropriate instrumentation can provide reliable knowledge of machine condition and allows relatively inexpensive repair on a prescheduled controlled basis in comparison to catastrophic failures that may cost twenty to thirty times more than the original job.

Rolling element bearings are subjected to wear during their operation and as a result their theoretically prescribed geometrical dimensions, their trueness of form and surface quality change, causing an increase in the vibration generated by bearings. In a geared rotor, the vibration caused by bearings are coupled with those due to gears, making the fault detection and monitoring more complex. Analysis of the vibration signals, both in low and high frequency regimes, for detection of bearing faults, becomes essential.

Low frequency vibrations are generally characteristic of the whole machine, mostly imbalance and gearing. Only very advanced bearings can be detected by sensing low frequencies. The need for frequency range (>5 khz) arises for better characterization of bearing faults.

1.1 LITERATURE SURVEY

The use of vibration data from rotating machinery to determine machine health has a long history. Traditionally a shop foreman or maintenance operator would press the end of a screwdriver against the bearing housing and the handler against his ear, or rather his mastoid, and in this way conduct the pure vibration signal from the bearing to be analyzed to his brain.

The way forward lay with the study of vibration signatures and frequency analysis with the advent of modern F.F.T. analyzers and now very powerful computerized techniques to analyze the spectrums. Levate [1] studied the changes in vibration signature of the bearings in good condition and the same bearing with dents in outer race. He reported that the change in this signature indicated

incipient failure , long before any external signs become evident. Mitchell and Lynch [2] studied the vibration of a gear pair and tried to identify the origin of noise. They reported that in the frequency domain there may be some components which might not be understood related to the gear system. These components are associated with inaccuracies manufactured into the gear. During the manufacture erroneous table positioning relative to the gear cutter results in gear errors. In effect , the cutting machine generates a ghost gear on the actual gear.

Lipovszky [3] reports a comprehensive method for the testing of rolling element bearing by frequency analysis method and gives a detailed reasoning for analysis manually in lower and higher frequency ranges.

Notes on signal analysis and instrumentation by Bruel & Kjaer [4] give methods for using various equipment like analyzers , pickups , amplifiers etc and further deals with signal analysis. Martin Angelo [5] reported analysis of vibration spectra keeping in view all kinds of mechanical faults and giving a systematic method for analyzing spectra in different frequency ranges. Application notes

by Bruel & Kjaer [6 & 7] also give formulae for calculations of impact rates for bearing components , methods of data acquisition , use of equipment & instrumentation.

Randal [8] in his book "Frequency Analysis" presented a comprehensive study of the signal analysis in the frequency domain. Campel [9] discussed the application of passive diagnostics and reliability experiments in machine condition monitoring.

A very recent paper by Braun & Snear [10] investigates the accuracy of spectral estimates computed via parametric techniques. He also compares the results obtained via classical Fourier method. Another very recent paper by Gu-Song-Nian & Zhang-Gian-Lin [11] discusses a vibration approach to structural faults where the sensitivity of structural vibration parameters to fault is analyzed.

1.2 PRESENT STUDY

Present study concerns itself with fault diagnosis of rolling element bearings in a geared rotor. An experimental test rig is fabricated

and installed. Vibration signatures for bearing defects like play between races, missing rolling elements, free rolling elements are obtained and compared with those obtained from undamaged bearings. Three cases of defective ball bearings are studied. Spectral techniques are employed. Analysis is carried out in the low, medium and high frequency zones. Changes in the vibratory response spectra a associated with the gravity of bearing faults.

A brief review of vibration of rolling element bearings is presented in chapter 2. Chapter 3 deals with the experimental investigation carried out. Chapter 4 concludes and discusses about limitations and scope for future.

CHAPTER 2

VIBRATION OF ROLLING ELEMENT BEARINGS

An ideal rolling element bearing (without defect) produces vibrations due to the clearance between the inner and outer race centers. The center of the inner race keeps changing depending on the position of the ball bearings. Consider the inner and outer race configuration of Fig 2.1.

The center of the outer ring is O_k. The inner ring is displaced, because of the radial clearance, in the direction of the load and its center moves to O_{b1}. This displacement equals half the radial play if the center of the rolling body falls on the line of action of the load. If two adjacent rollers are positioned symmetrically with respect to the line of the acting force, then the inner ring, supported on both rollers, is displaced further and its center moves to O_{b2}. The position of the center of inner ring will change periodically between O_{b1} and O_{b2}, depending on the position of the

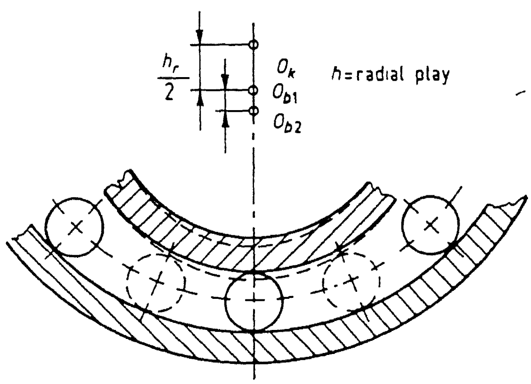

FIG 2.1 VIBRATIONAL MOTION OF THE INNER RING OF A ROLLING BEARING

rolling bodies. Half the distance between O_{b1} and O_{b2} is the amplitude of the vibrational motion and its frequency is

$$f = 1/120(1-\gamma)\ Z \times n \text{ Hz},$$

where

$\gamma = d_g \cos \alpha / d_m$;

d_g = diameter of rolling body (mm);

α = contact angle of the bearing;

d_m = mean dia of the bearing(mm);

Z = number of rolling bodies running in one line;

n = number of revolutions(1/min).

The twofold theoretical amplitude of the vibration of the inner ring can be calculated from the geometrical data of the bearing by means of the following relationship [12]:

$$\Delta = (d_m/2+h_r/4)\cos(\pi/Z)-[(d_m/2+h_r/4)\cos(\pi/Z)^2-d_m\ h_r/2]^{1/2}-h_r/2.$$

2.1 VIBRATION CAUSED BY BEARING FAULTS

Rolling bearings are subjected to wear during their operation, as all are the machine parts, and as a result their theoretically prescribed geometrical dimensions, their trueness of form and surface quality change, causing a change, usually an increase, in the vibration generated by the bearings. As the inner and outer rings and the rolling bodies in ball bearings and cylindrical roller bearing wear, the radial play increases.

As a consequence the rotating shaft is displaced from its original axis, and its rotation then no longer continues to be about the geometrical center. The changes caused by wear lead to further displacement of the bearing components from each other and increased mutual impacts, resulting in enhanced vibrations. The irregularly worn and damaged rolling bodies will no longer move smoothly and regularly in the live rings which have irregular, worn and damaged shapes.They will impact against each other, raising vibrationl waves depending upon the amount of wear and damage and their motion relative to each other.

Generally the vibrations raised by impacts are gradually damping, transient vibrational phenomena, but they occur periodically, depending on the relative speed arising because of the rotation of the bearing. Therefore, the number of repeated impacts causing the vibration and hence the frequency of the shock waves are determined by the speed, the number of rolling bodies, and other factors.

The impact processes occur, when a rolling body rolls on an outer ring of infinite radius until it arrives at a damaged part of the track.

If the track and the rolling body had perfect ideal shapes and ideal surfaces , the latter would roll smoothly along the track without any interruption. But a rolling element bearing containing a fault (typically a crack or corrosion pit) either on the inner race or the outer race or on the rolling element itself or the play between inner and outer race. This crack will create small impulses every time one of the rolling element passes over it. These impulses like a series of hammer blows , will transmit energy to the bearing housing which in turn will vibrate at its natural frequency and decay with the damping in mechanical structure

similar to striking a brass bell. The structure acts in this way as a mechanical amplifier.

Local impact forces are also created at the points of contact depending on the speed of impact. Compression waves propagate in the material at the velocity of sound which is characteristic for the material. In the next phase of impact the track and rolling bodies are deformed and part of energy of impact is dissipated , inducing vibrations. Therefore the whole bearing performs a complex motion.

Also the vibration caused by the impact is complex : It consists of vibrations of a relatively low frequency which are produced by deformations of the track and a high frequency vibration of about up to 40 khz super imposed on this.This can be understood better if we regard the track as a beam (fig.2.2) with two supports. The beam is bent by ball falling on it and performs a damped vibration with relatively low frequency. The magnitude of the deflection depends on the energy of the ball falling on the beam. High frequency vibrations and shock waves emanate from the point of impact of the ball on the beam.

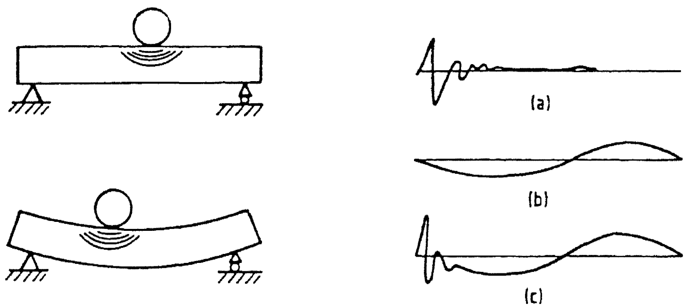

FIG 2.2 WAVES FORMED DURING AN IMPACT
(a) and (b) components; (c) resultant

The transient shock wave of the impact is formed in a very short time causing very rapid vibrations in the body of bearing. The frequency of the resulting waves and their damping depend on the quality of the bearing and of the material.

When testing a bearing the mutually superimposed vibrations are separated. In the start up phase, a bearing fault is hard to recognize from observations of the low frequency vibrations, because of disturbing vibrations from other sources, while even quite insignificant bearing faults can be observed easily by studying the high frequencies.

2.2 VIBRATION ANALYSIS IN THE FREQUENCY DOMAIN

The previous discussion pertain to individual and isolated cases of faults in bearings. Faults in bearings may however occur simultaneously (say on a ball as well as one/both races). Moreover bearings would be one of the numerous machine elements in an installation and vibration due to bearing defects would be invariably mixed with those from the other sources on the installation. The vibration signal sensed on the installation would

therefore, comprise of a number of frequency components.

It is therefore imperative that the analysis be carried out in the frequency domain and frequencies characterizing bearing vibration be identified and monitored for incipient faults, on the vibration spectrum.

The frequency spectra can be divided into the following ranges:

a) low frequency range.(0-250 hz)

b) medium frequency range.(250 hz-2.5 khz)

c) high frequency range.(2.5 khz-25 khz)

2.2.1 LOW FREQUENCY RANGE

The low frequency components in a spectrum carry the shaft revolution frequency and its harmonics. Vibration at these frequencies generally arise due to unbalance, misalignment and shaft bending.

In addition it also carries information at bearing impact rates. These impact rates are the frequencies which are function of rotor

speeds and as discussed earlier come from the impact of the ball on the individual components like inner race , outer race or ball itself. The vibration levels at these frequencies are associated with shocks depending upon the gravity of the faults present in the individual component.

The impact rates as discussed in Bruel & Kjaer application notes are given by ;

1) for outer race defect

$f(hz) = n/2\ Fr\ (\ 1-BD/PD\ COS\ \beta\ )$

2) for inner race defect

$f(hz) = n/2\ Fr\ (\ 1+BD/PD\ COS\ \beta\ )$

3) for ball defect

$f(hz) = PD/BD\ Fr[\ 1-(BD/PD\ COS\ \beta\ )^2\ ]$

where

n = number of balls or rollers

Fr = relative rev/sec between inner and outer races

Monitoring the impact rate frequencies, their harmonics and the vibration level at these frequencies can provide clues about faults in the individual components.

2.2.2 MEDIUM FREQUENCY RANGE

In the medium frequency range, the components are mainly associated with vibrations originating from the tooth mesh in the gear box These components occur at frequencies identified as tooth meshing frequency and its harmonics. The tooth meshing frequency is defined as:

Tooth Meshing Frequency = m (r.p.m/60) k

m = harmonic number

k = number of teeth

Bearing defects in this range are shown in the form of increase in vibration level at these frequencies since the energy dissipation due to bearing faults is shared by the gear-box reflecting in a rise of amplitude levels at tooth meshing frequencies.

The natural frequencies of the system may also lie in the medium frequency zone. These frequencies also need to be monitored for vibration amplitude since a bearing defect between balls and races gives rise to an impact, setting the system into free vibrations.

2.2.3 HIGHER FREQUENCY RANGE

The higher frequency zone contains information about the higher natural modes of installation. The rolling element, on encountering a fault, sends a shock pulse to the installation and sets it into free vibration which occurs at its natural frequencies.

The lower mode natural frequencies of the installation may lie in the medium frequency range and the spectrum at these frequencies may get shadowed by the other spectral properties of the range (e.g. tooth meshing frequencies) rendering it very complex for identification and analysis.

The higher frequency range however contains information only about the higher natural frequencies and makes it easy to detect occurrence of transient free vibration.

CHAPTER 3

EXPERIMENTAL INVESTIGATIONS

An experimental rig was setup in the laboratory to carry out investigations on vibration of rolling bearings. Fig 3.1 gives the schematic layout of the setup. The rig consists of two parallel shafts each supported on a set of identical ball bearings. The ball bearings are supported in bearing housings mounted on a base plate, designed for the purpose, which in turn is bolted on a heavy wooden table.(Figs 3.2-3.4) One of the shafts is coupled through a flexible coupling to a D.C. motor and drives the other shaft through a spur-gear arrangement. The driven shaft carries a loading drum at its other end.A rectifier is designed to be used along with the D.C motor to convert the A.C. power supply to D.C. A variac was used to control the input voltage to the rectifier and hence the rotational speed.(Fig 3.2).

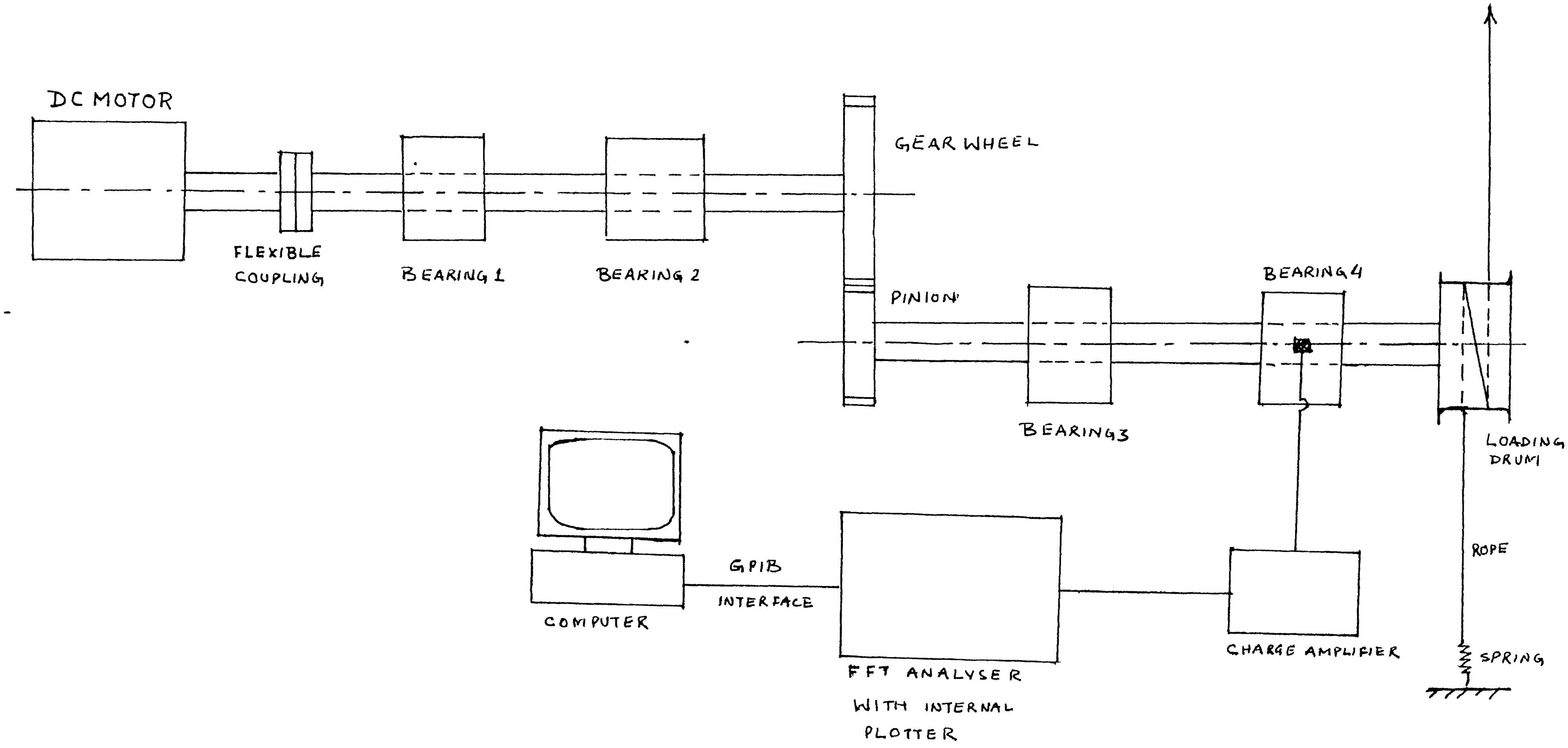

FIG 3.1 SCHEMATIC LAYOUT OF THE TEST RIG

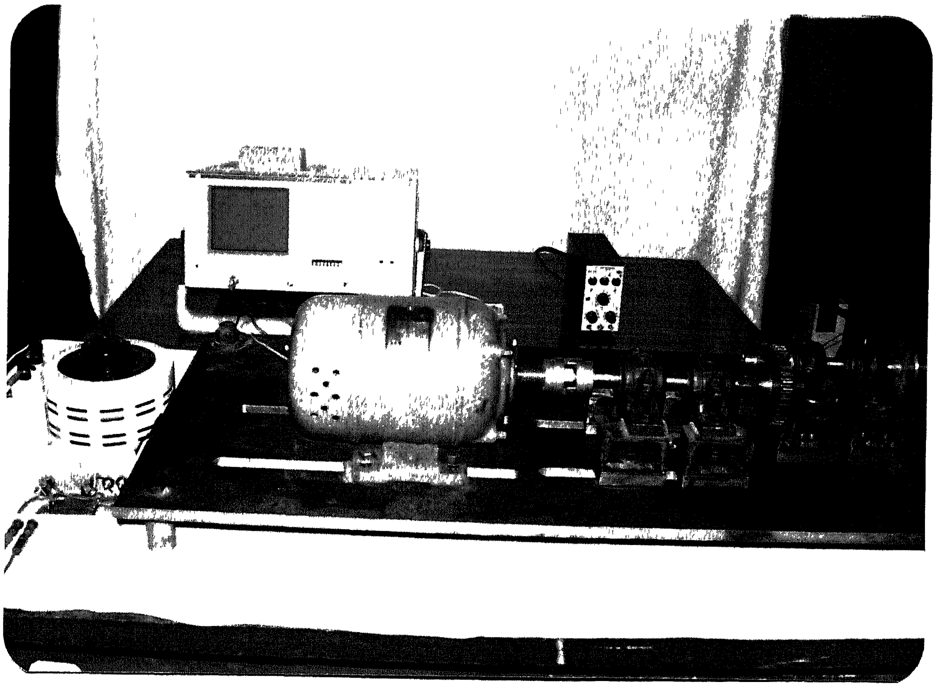

FIG 3.2 EXPERIMENTAL SETUP

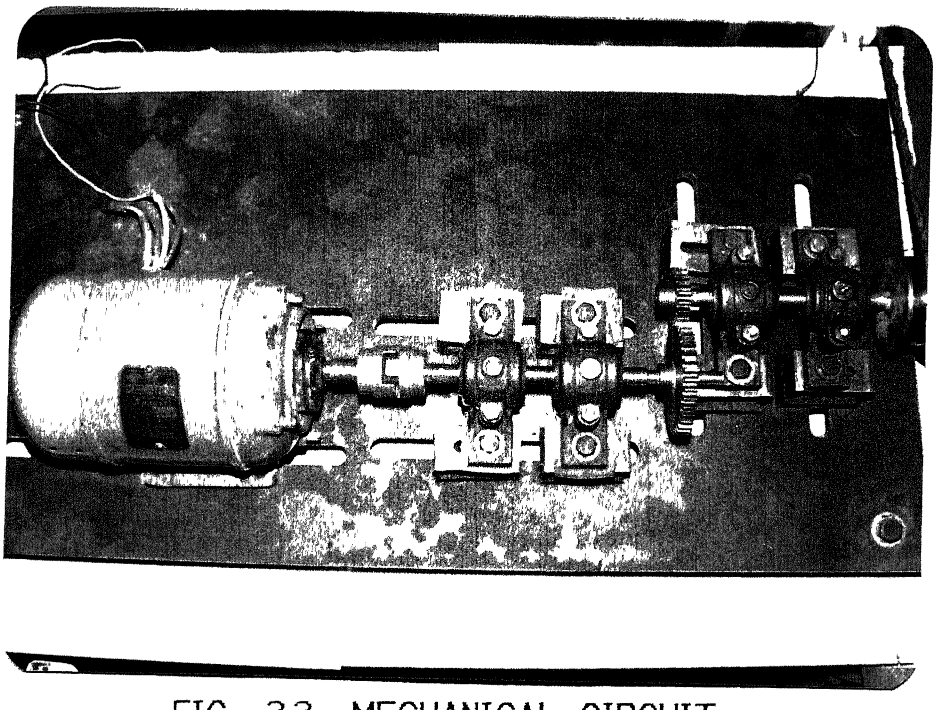

FIG 3.3 MECHANICAL CIRCUIT

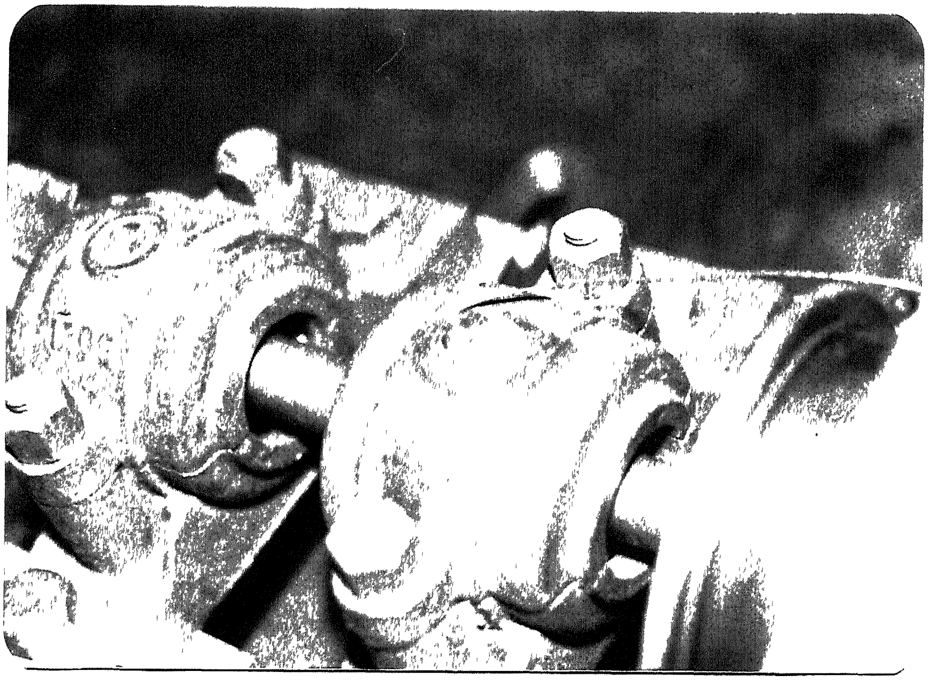

FIG 3.4 ACCELEROMETER MOUNTING

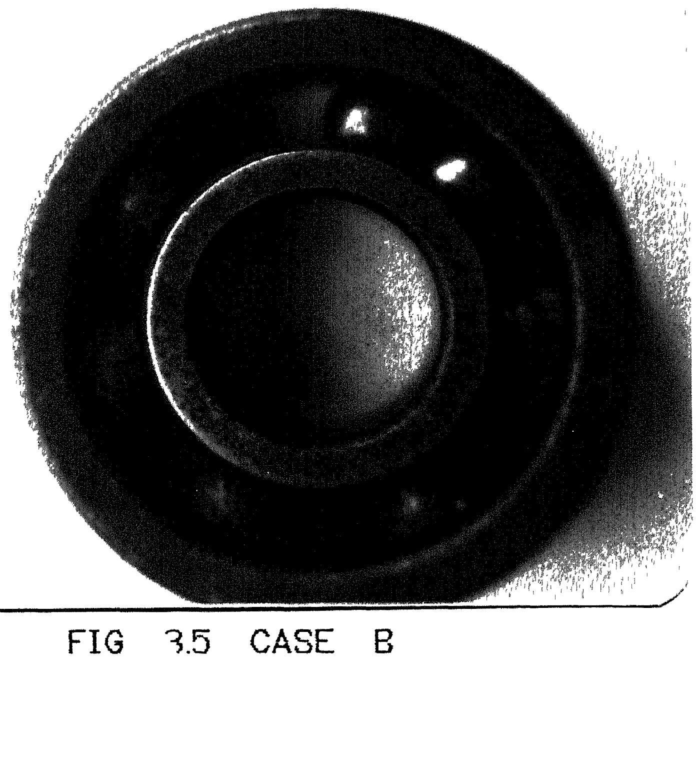

FIG 3.5 CASE B

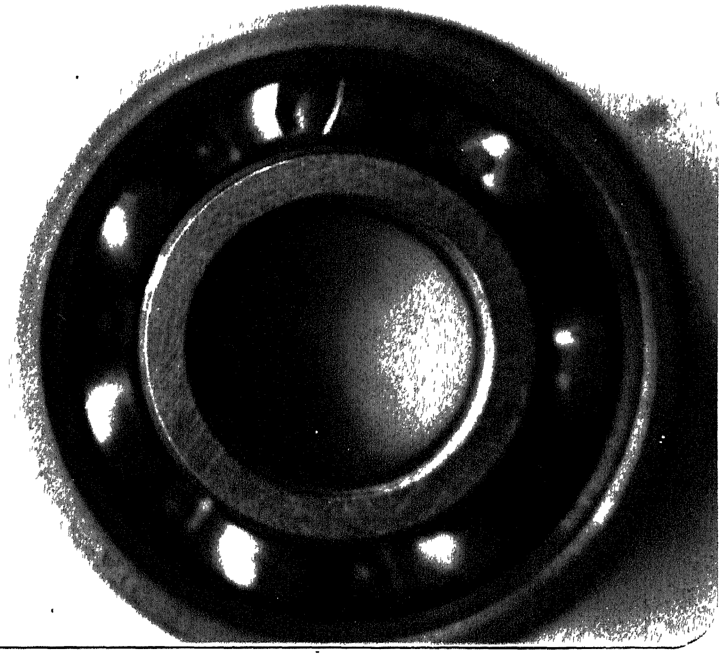

FIG 3.6 CASE C

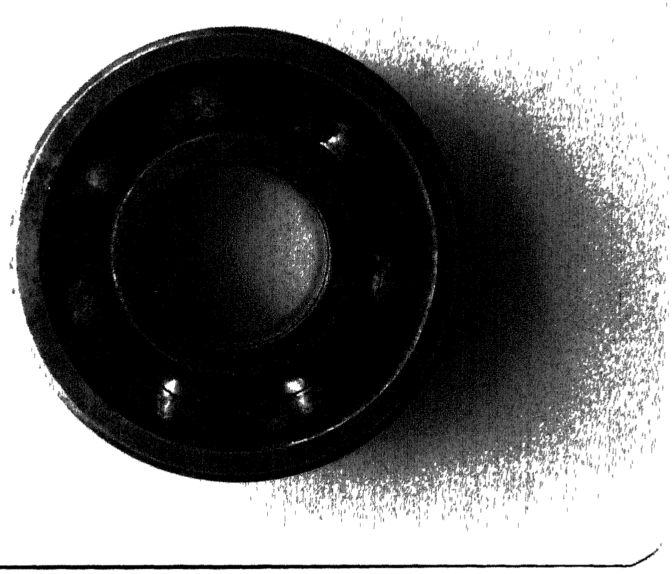

FIG 3.7 CASE D

3.1 INSTRUMENTATION

The vibrations were picked up by an accelerometer mounted on one of the bearings. The bearings were identified as bearing 1, bearing 2, bearing 3 and bearing 4 (Figs 3.5-3.7) for the purpose of reference. Bearing 4 was chosen as the sensor location and the accelerometer was mounted on its housing with the help of epoxy cement. The signals from accelerometer were amplified in an amplifier and fed to a digital storage oscilloscope , capable of performing F.F.T on the incoming signal. The oscilloscope was connected to a computer through a GPIB interface.The details and specifications of the instruments employed are given in appendix 1.

3.2 IDENTIFICATION OF NATURAL FREQUENCIES

Prior to obtaining the vibration signatures of healthy and defective bearings, natural frequencies of the system were identified. This was done to distinguish this set of frequencies from those arising due to rotation of the shafting. The natural frequencies were identified by performing a Rap Test on the installation in stationary condition.

One of the bearing locations, where the defective bearing is to be introduced later (bearing 1, Fig3.1), is struck with a light wooden hammer to set the installation into free vibrations. The frequency spectrum of the free vibrations identifies the natural frequencies.

Figs 3.8(a) and (b) are the natural frequency spectra plotted in frequency ranges 0-10 khz and 0-25 khz respectively.

3.3 CASE STUDIES

The objective of the experimental investigations is to compare the spectra of defective bearings with that obtained from a healthy bearing to identify the nature and severity of the fault. The vibration spectra is first obtained for a set of healthy bearings, at a number of rotor speeds. These spectra are employed as a datum for comparison. One of the four bearing position is identified for introduction of faulty bearings, one by one. The location of the sensing accelerometer is kept fixed in all cases.

The experimental exercise was constrained in the introduction of faulty bearings. Introduction specific, individual faults in

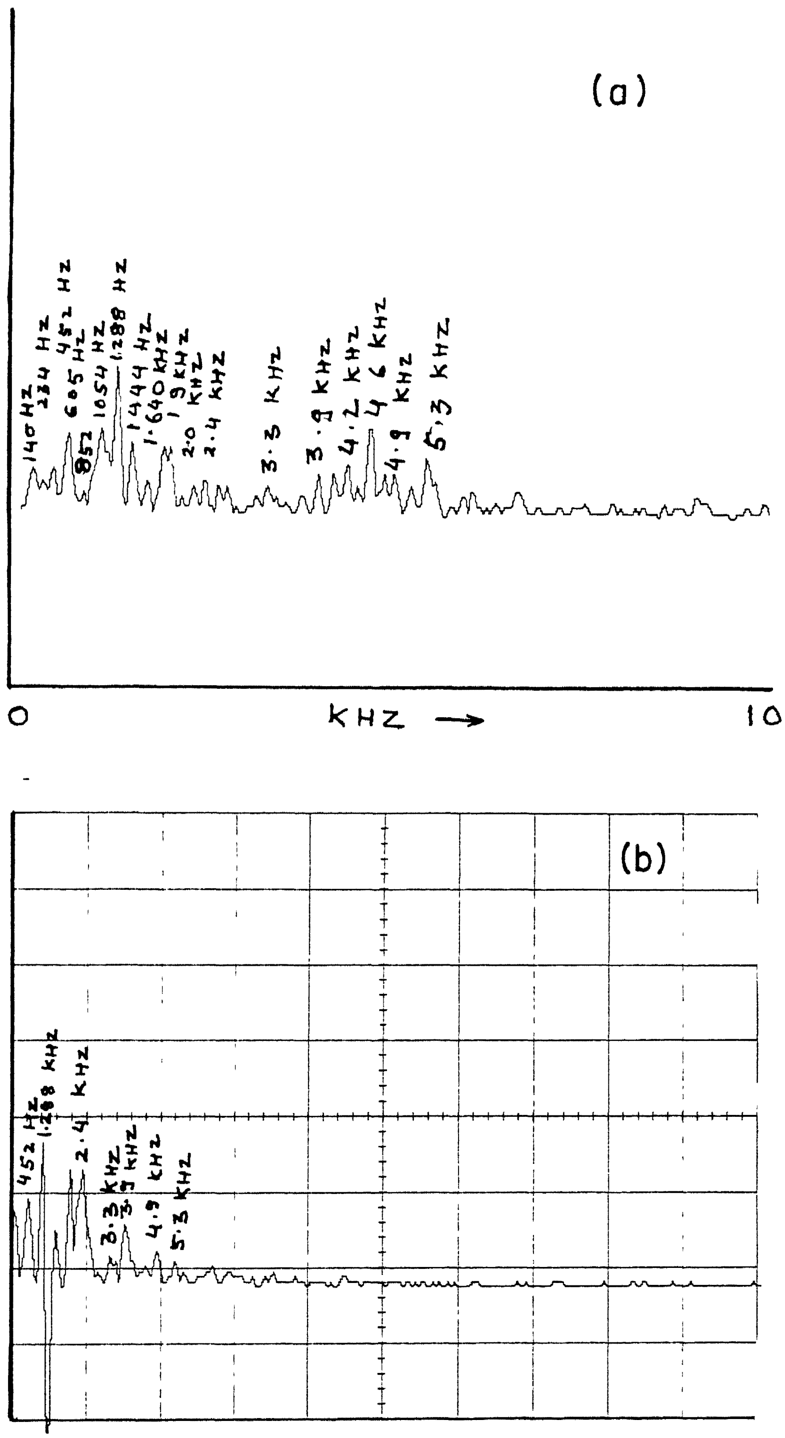

FIG 3.8 NATURAL FREQUENCIES

bearings(e.g. a crack in the outer race or crack in the ball) was not possible within the available workshop machinery. The defective bearings were, therefore produced from National Engineering Inds., Jaipur. However, again it was not possible to generate individual and isolated faults in bearings and defective bearings were selected for different order of severity of defects.

The case studies undertaken are

[A] All bearings healthy.

[B] Bearing 1 with broken cage, worn out races and two balls free.

[C] Bearing 1 with a damaged ball.

[D] Bearing 1 with excessive play between inner and outer races.

Figs 3.5-3.7 show the defective bearings.

The investigations are carried out at four rotational speeds:

1) 400 r.p.m

2) 600 r.p.m

3) 800 r.p.m

4) 1000 r.p.m

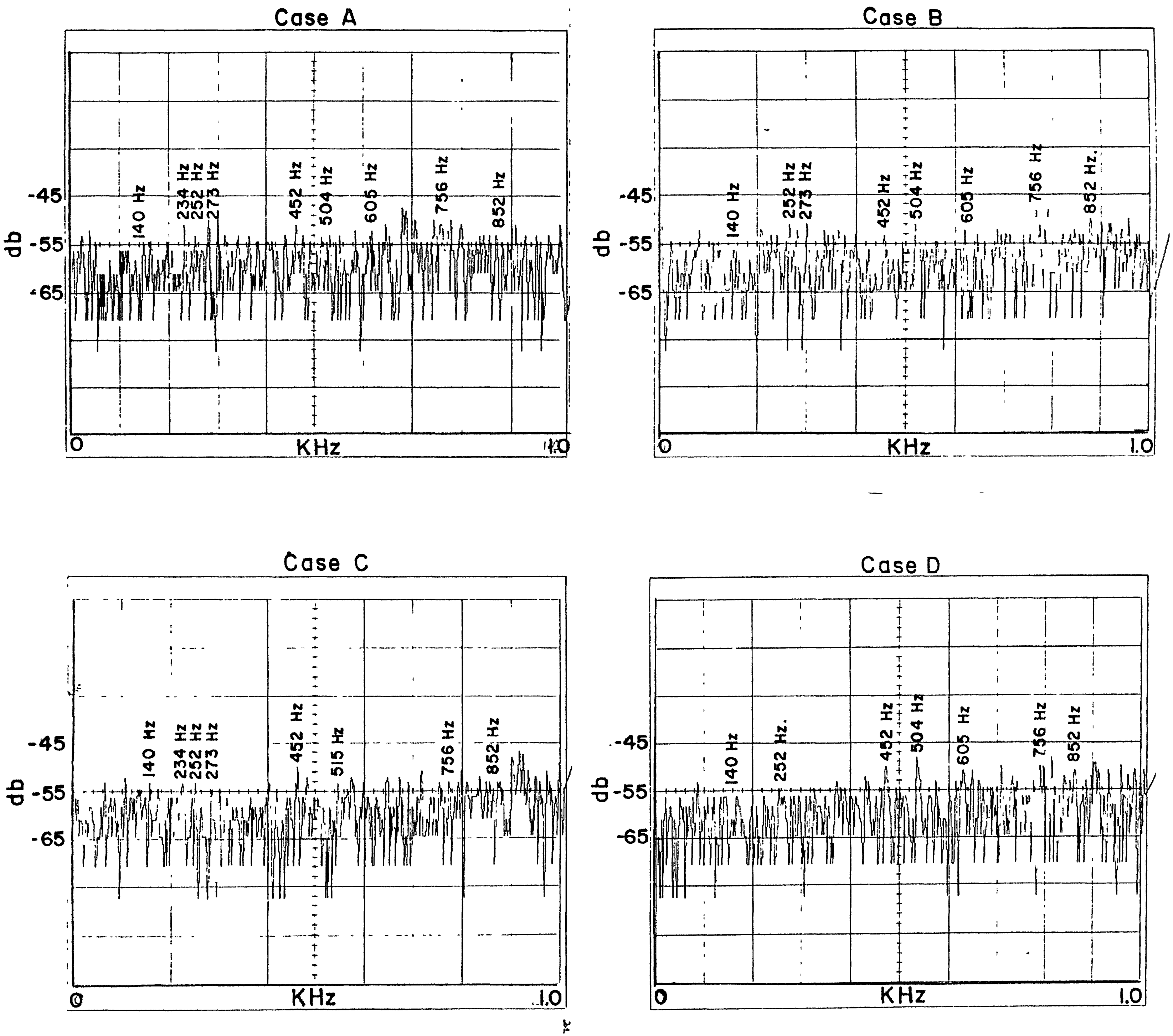

Fig 3.9 FREQUENCY SPECTRUM AT 400 R.P.M(MEDIUM FREQUENCY RANGE)

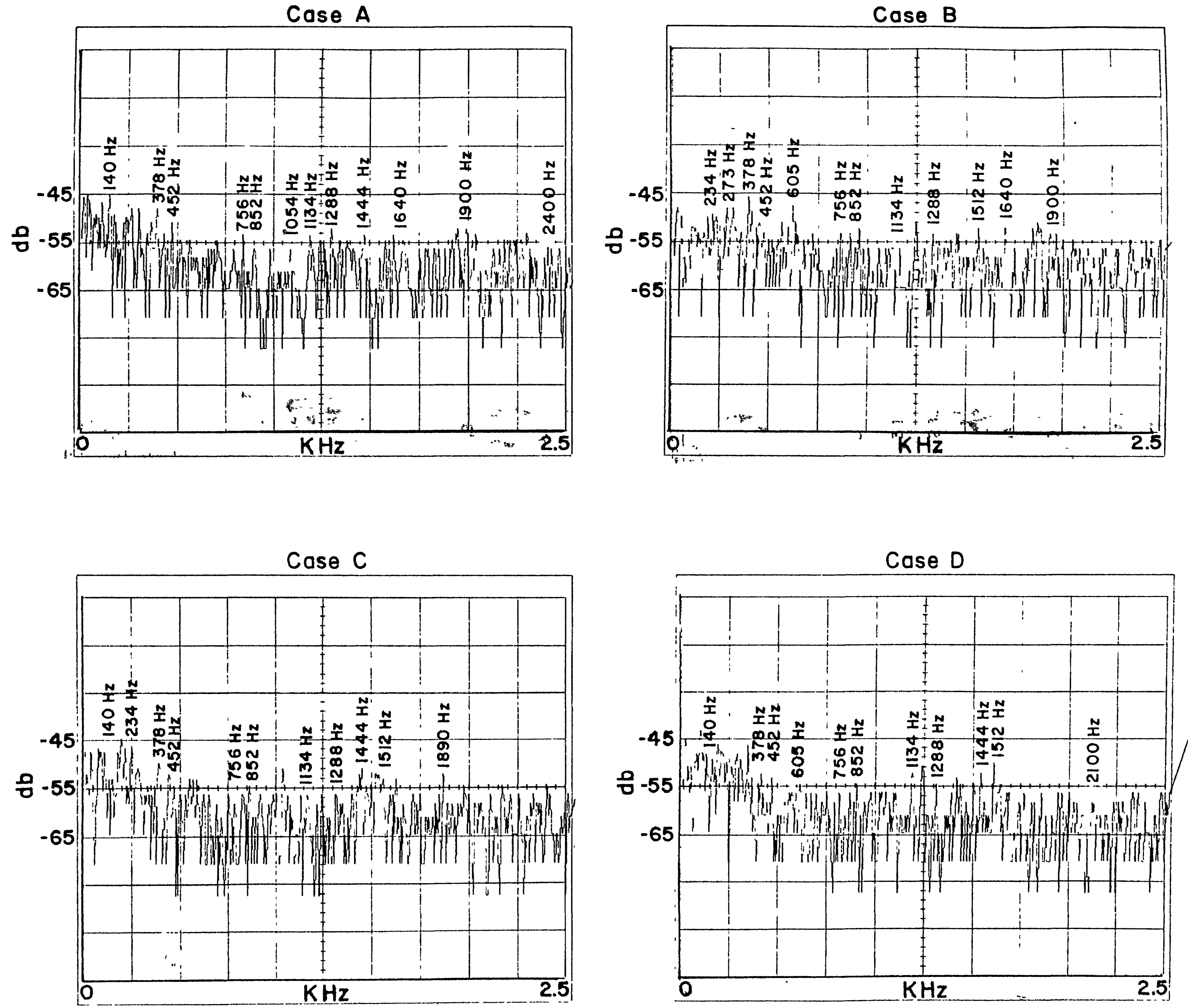

Fig 3.10 FREQUENCY SPECTRUM AT 600 R P M(MEDIUM FREQUENCY RANGE)

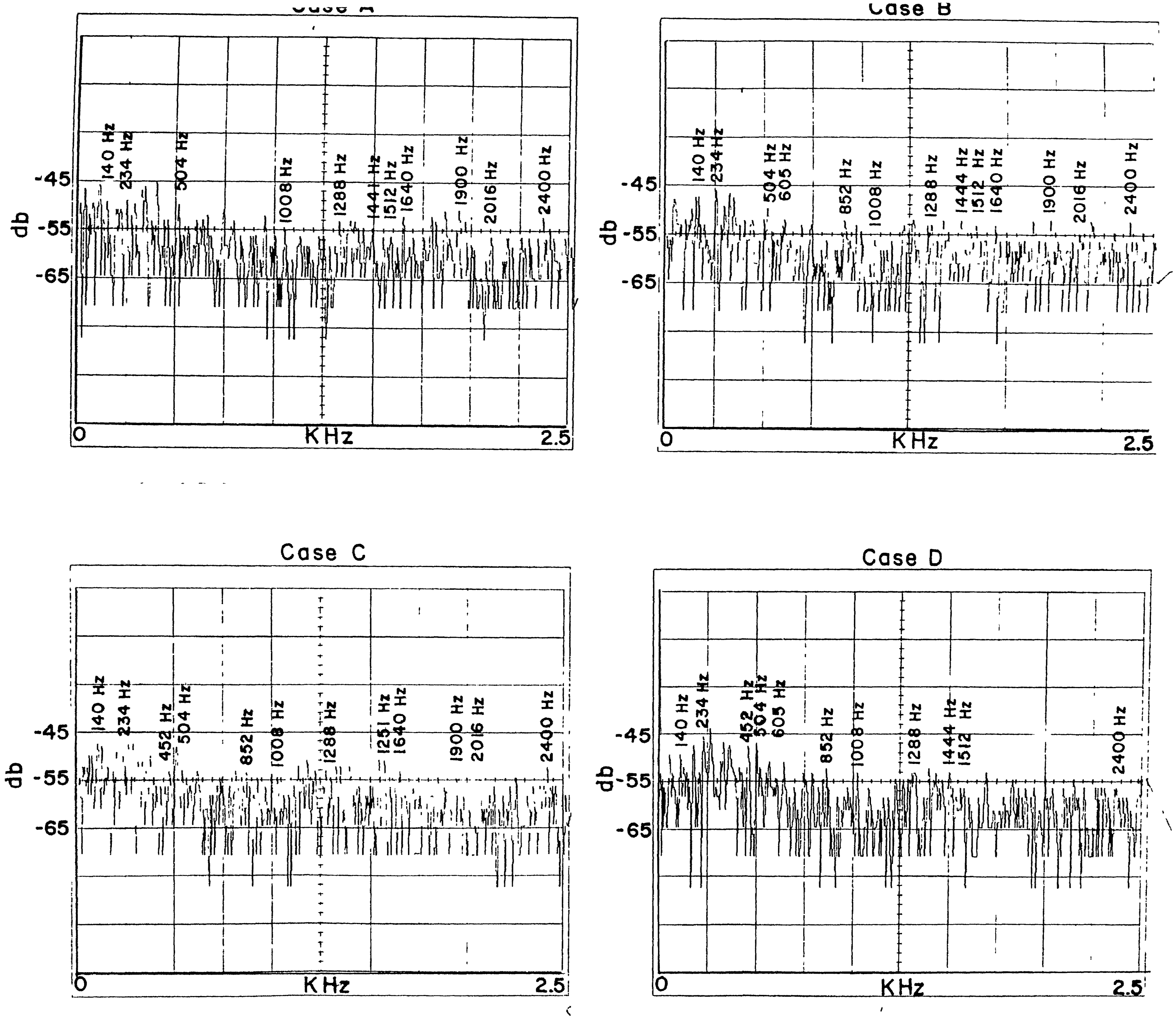

Fig 3.11 FREQENCY SPECTRUM AT 800 R P M(MEDIUM FREQUENCY RANGE)

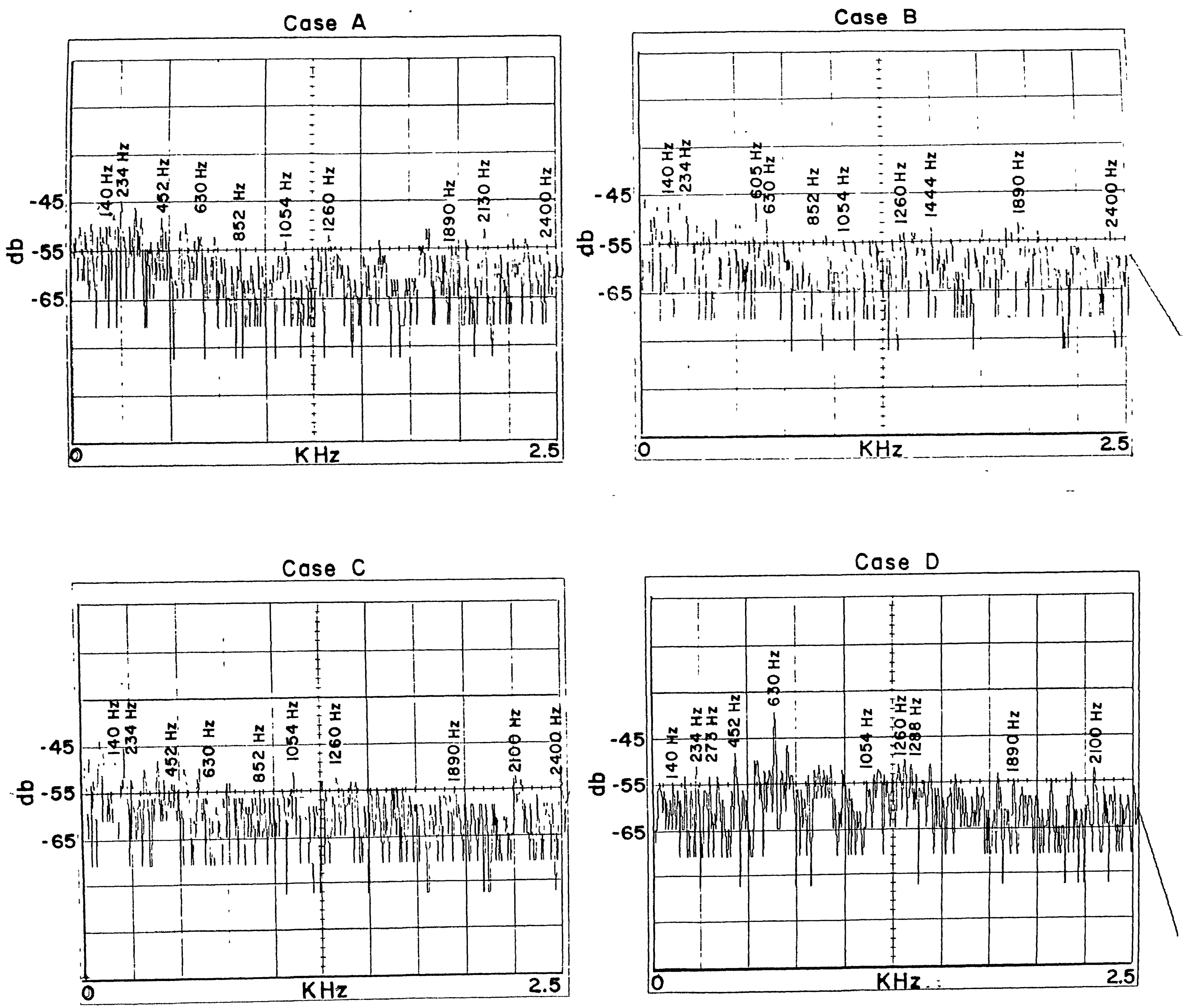

Fig 3.12 FREQUENCY SPECTRUM AT 1000 R P M (MEDIUM FREQUENCY RANGE)

The spectral analysis is carried out in following frequency ranges:

a) 0 to 100 hz / 0 to 250 hz (low freq range)

b) 0 to 1 khz / 0 to 2.5 khz (medium freq range)

c) 0 to 25 khz (high freq range)

3.4 LOW AND MEDIUM FREQUENCY ANALYSIS

Figs 3.9 to 3.12 are the spectra for the four bearing cases under consideration for rotor speeds of 400 rpm, 600 rpm, 800 rpm and 1000 rpm. These spectra are shown in the frequency range 0-1 khz/2.5 khz and contain information about the zones identified as low and medium frequency ranges. (High frequency spectra are given later). Ideally for case A, the spectra with healthy bearing should contain harmonics tooth meshing frequency only arising out of tooth loading at meshing frequency rate given by:

T.M.F =m (r.p.m/60) k

m = harmonic number

k = number of teeth

However, complexities in the spectra can be observed due to following reasons.

1) Due to unavoidable machining errors , the tooth profile may not be ideal and also teeth may not be identical to one another causing variations in tooth deflections at harmonics of rotational frequencies leading to modulation between tooth meshing and rotational frequency harmonics.

2) The test rig is not perfectly aligned , giving rise to components of vibration at 1xr.p.m , often 2xr.p.m and sometimes 3 & 4xr.p.m.

3) Residual unbalance in rotating members causing vibration at 1xr.p.m.

4) Electrically induced vibration can be seen at 1xr.p.m or 2xsynchronous frequency.

5) Even the new rolling bearings give rise to vibrations because of bearing clearance , waviness of races , ovalness of balls and intrusion of foreign particles.These components of vibration can be seen in lower as well as higher frequency range. Also , the motor bearings, which are not new, give rise to vibrations which can be seen in vibration spectra.

6) GHOST COMPONENTS: These are the components which could not be related to any of the above discussed phenomenon and thus will be called as ghost components and are a result of accidental peaks.

The points of interest in the frequency spectra are characterized as follows:

a) Impact Rates: Impact rates at all r.p.m's were calculated and are tabulated in table 3.1.

b) Tooth Meshing Frequencies: Tabulated in table 3.2.

c) Natural Frequencies: Obtained earlier are tabulated in table 3.3.

These are monitored for trends on introduction of faults in bearings. The vibration amplitudes at these frequencies for the four cases at various rotor speeds are compared in tables 3.1 to 3.3. Besides these points , in high frequency range (0-25 khz) spectra for faulty bearings are compared with that of new bearing for detection of faults.

3.4.1 IMPACT RATES :

Table 3.1 summaries the vibration characteristic at impact rate frequencies. A marked rise in the amplitude level at these points can be clearly seen at all rotational speeds, for all three defective cases. The vibration amplitudes at these frequencies are

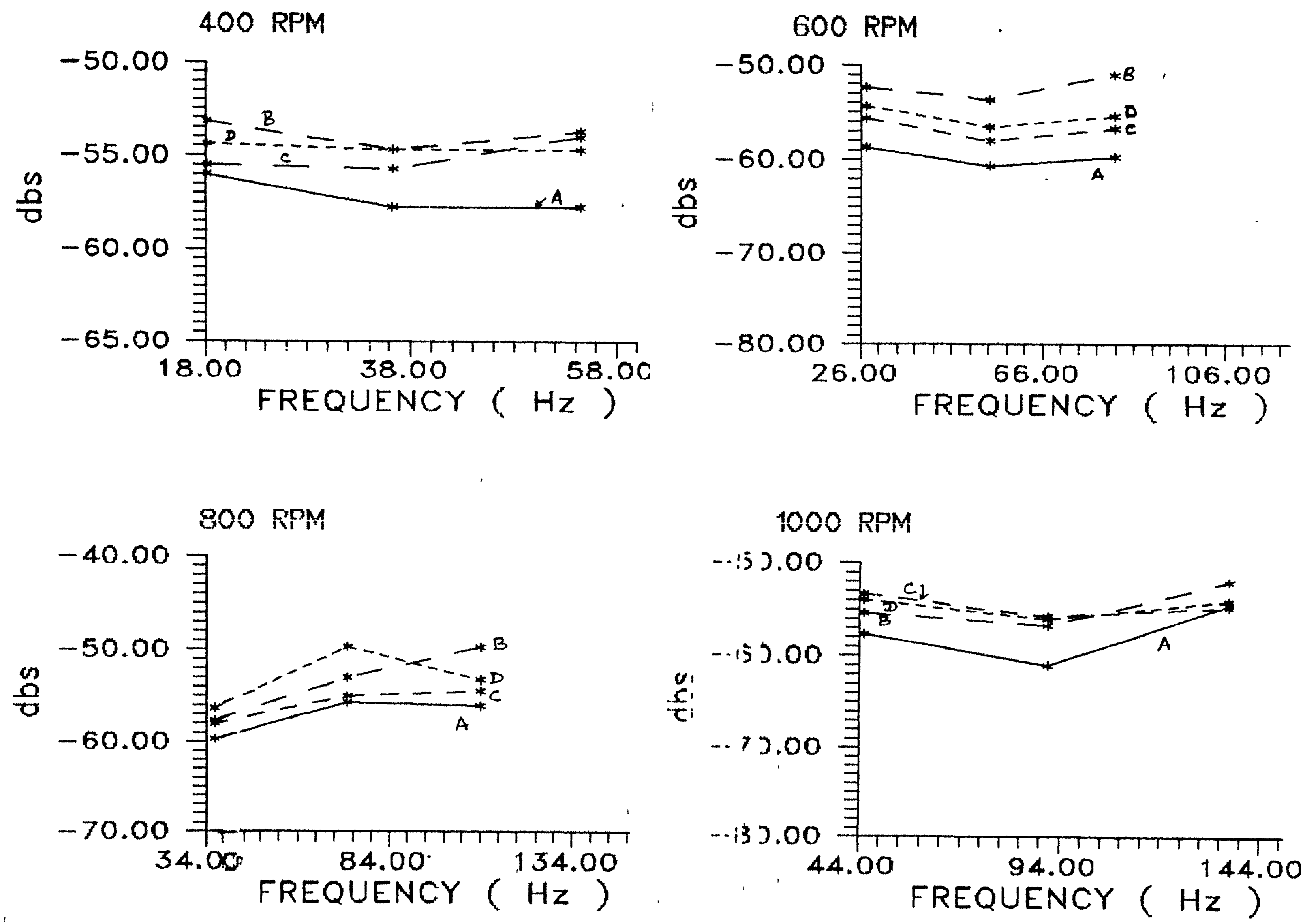

Fig 3.13 AMPLITUDE LEVELS AT IMPACT RATES
(FOR OUTER RACE)

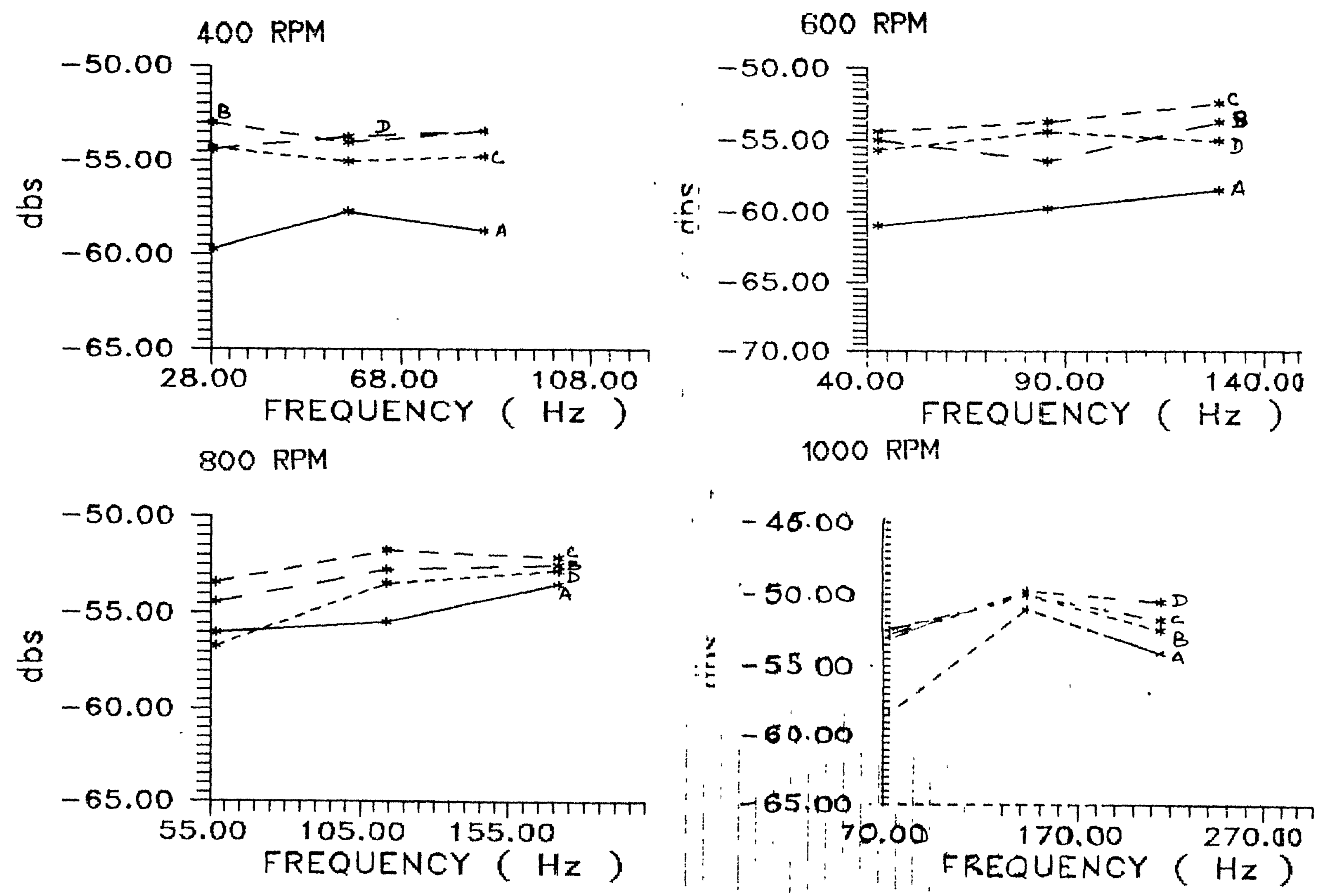

Fig 3.14 AMPLITUDE LEVEL AT IMPACT RATES
(FOR INNER RACE/BALL)

also plotted in Figs 3.13-3.14. The rise in the amplitude levels is dependent upon the gravity of fault.

For case B, where fault is very complex (cage broken , two balls free & play between races) , the rise in vibration level is more as compared to case C, where only one ball is damaged. (fig 3.13). However , this rise is less in case D (excessive play between inner and outer races) . For cases B & D , almost uniform rise in vibration levels can be observed at impact rates for outer race , inner race & balls; For case C, the increase in vibration level is more for impact rates related to balls than for races. This can be explained by the fact that , for cases B & D, all three components are faulty whereas for case C only one ball is damaged.

3.4.2 TOOTH MESHING FREQUENCIES

Table 3.2 gives the amplitude levels at tooth meshing frequencies for all four cases , at all rotational speeds. For case C rise in vibration level is less then cases B & D where faults are more grave. Fig 3.15 contain plots for all the four cases for comparison. As can be seen, plot for case C lies below plots for cases B & D.

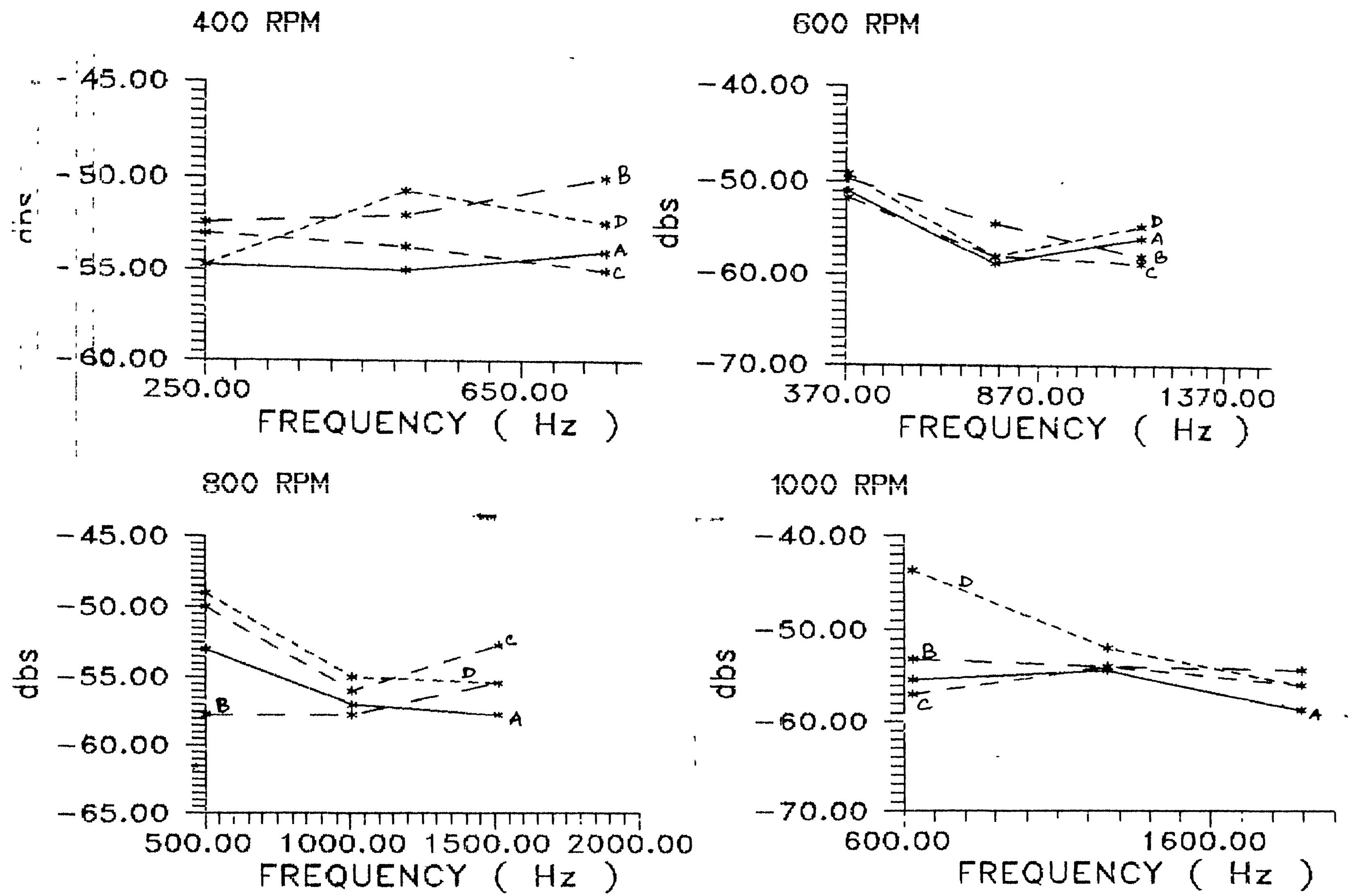

Fig 3.15 AMPLITUDE LEVELS AT TOOTH MESHING FREQUENCIES

TABLE 3.1

IMPACT RATES

TABLE 3.1(a) (400 RPM)

IMPACT RATE (HZ)	AMPLITUDE (db)			
OUTER RACE				
	CASE A	CASE B	CASE C	CASE D
18.13	-56.0	-53.2	-55.0	-54.4
36.30	-57.7	-54.7	-55.7	-54.7
54.40	-57.7	-53.7	-54.0	-54.7
INNER RACE/BALL				
28.5	-59.7	-54.4	-53.0	-54.3
57.0	-57.7	-53.7	-54.0	-55.0
85.5	-58.7	-53.4	-53.4	-54.7

TABLE 3.1(b) (600 RPM)

IMPACT RATE (HZ)	AMPLITUDE (db)			
OUTER RACE				
	CASE A	CASE B	CASE C	CASE D
27.22	-58.7	-52.4	-55.7	-54.4
54.44	-60.7	-53.7	-58.0	-56.0
81.66	-59.7	-51.0	-56.7	-55.4
INNER RACE/BALL				
42.77	-61.0	-55.0	-54.4	-55.7
85.54	-59.7	-56.4	-53.7	-54.4
128.31	-58.4	-53.7	-52.4	-55.0

TABLE 3.1(c) (800 RPM)

IMPACT RATE (HZ)	AMPLITUDE (db)			
OUTER RACE				
	CASE A	CASE B	CASE C	CASE D
36.28	-59.7	-57.7	-58.0	-56.4
72.56	-55.7	-53.0	-55.0	-49.7
108.84	-56.0	-49.7	-54.4	-53.2
INNER RACE/BALL				
57.02	-56.0	-54.4	-63.4	-56.7
114.04	-55.4	-52.7	-51.7	-53.4
171.06	-53.4	-52.4	-52.0	-52.7

TABLE 3.1(d) (1000 RPM)

IMPACT RATE (HZ)	AMPLITUDE (db)			
OUTER RACE				
	CASE A	CASE B	CASE C	CASE D
45.35	-57.7	-55.4	-53.4	-54.0
90.70	-61.0	-56.7	-55.7	-56.0
136.05	-54.4	-52.0	-54.7	-54.0
INNER RACE/BALL				
71.26	-58.2	-53.0	-52.7	-52.4
142.52	-51.0	-49.7	-50.0	-50.0
213.78	-54.0	-50.4	-51.7	-52.4

TABLE 3.2

TOOTH MESHING FREQUENCIES

TABLE 3.2(a) (400 RPM)

TMF (HZ)	AMPLITUDE (db)			
	CASE A	CASE B	CASE C	CASE D
252	-54.7	-52.4	-53.0	-54.7
504	-55.0	-52.0	-53.7	-50.7
756	-54.0	-50.4	-55.0	-52.4

TABLE 3.2(b) (600 RPM)

TMF (HZ)	AMPLITUDE (db)			
	CASE A	CASE B	CASE C	CASE D
378	-51.0	-49.7	-51.7	-49.2
756	-58.7	-54.4	-58.0	-58.0
1134	-56.0	-58.0	-58.7	-54.7

TABLE 3.2(c) (800 RPM)

TMF (HZ)	AMPLITUDE (db)			
	CASE A	CASE B	CASE C	CASE D
504	-53.0	-57.7	-50.0	-49.0
1008	-57.0	-57.7	-56.0	-55.0
1512	-57.7	-55.4	-52.7	-55.4

TABLE 3.2(d) (1000 RPM)

TMF (HZ)	AMPLITUDE (db)			
	CASE A	CASE B	CASE C	CASE D
630	-55.4	-53.2	-57.0	-43.7
1260	-54.4	-54.0	-54.0	-52.0
1890	-58.7	-54.4	-56.0	-56.0

TMF--> TOOTH MESHING FREQUENCY

TABLE 3.3

NATURAL FREQUENCIES

TABLE 3.3(a) (400 RPM)

NAT FREQ (HZ)	AMPLITUDE (db)			
	CASE A	CASE B	CASE C	CASE D
140	-57.4	-54.4	-54.7	-56.7
234	-53.0	-54.7	-54.0	
273	-53.0	-54.7	-55.4	-55.4
452	-53.4	-55.0	-51.7	-52.0
605	-54.0	-53.4		-52.7
852	-56.4	-52.0	-55.4	-53.0

TABLE 3.3(b) (600 RPM)

NAT FREQ (HZ)	AMPLITUDE (db)			
	CASE A	CASE B	CASE C	CASE D
140	-48.7	-50.0	-54.0	
234	-54.0	-52.0	-49.7	-50.4
273		-51.0	-53.4	- 49.0
452	-54.2	-54.0		-54.0
605	-58.7	-50.0		-55.7
852	-57.7	-54.7	-56.7	-56.0

TABLE 3.3(c) (800 RPM)

NAT FREQ (HZ)	AMPLITUDE (db)			
	CASE A	CASE B	CASE C	CASE D
140	-52.7	-51.0	-53.0	-50.0
234		-49.4		
273			-50.0	
452	-55.4	-53.0	-56.0	-48.0
605	-55.4	-54.7	-54.7	-54.0
852	-58.0	-54.4	-55.0	-54.0

TABLE 3.3(d) (1000 RPM)

NAT FREQ (HZ)	AMPLITUDE (db)			
	CASE A	CASE B	CASE C	CASE D
140	-54.0	-47.7	-52.0	-50.0
452	-50.0	-52.7		-49.0
605	-56.0	-48.0	-52.0	
852	-56.4	-54.4	-55.0	-53.0

3.4.3 NATURAL FREQUENCIES

Table 3.3 summarizes the amplitude levels in db at natural frequencies. The rise in vibration levels at almost all these points for faulty cases appreciates the fact that the energy dissipated by the faulty bearing is also distributed to the free vibration energy of the installation at its natural frequencies. It can be observed that the rise in these amplitude levels correspond to the severity of the fault. However no conclusion can be made regarding the nature of fault.

It should also be noted that the spectrums are so rich and complex in frequencies, and monitoring vibrations is a difficult exercise the amplitude at natural frequencies are shown in Table 3.3 and it is clear that level for case C lies below that for cases B & D.

3.4.4 HIGH FREQUENCY RANGE

Bearing faults set a rotating installation into free vibrations due to the shocks they impart to the system. These transient vibrations occur at the natural frequencies of the installation. It is to be noted that rolling element bearings have high stiffness and

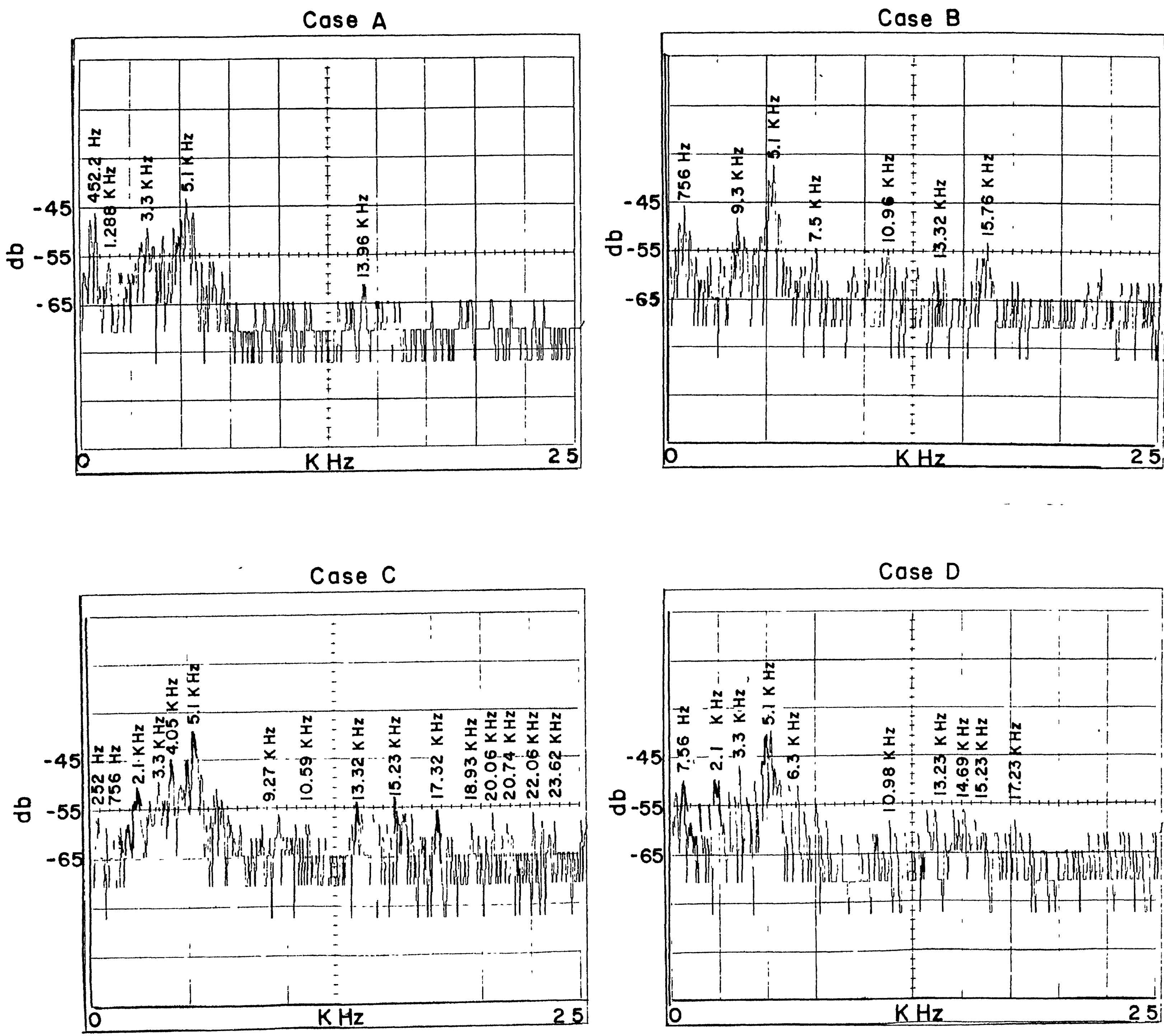

Fig 3.16 FREQUENCY SPECTRUM AT 400 R P M(HIGH FREQUENCY RANGE)

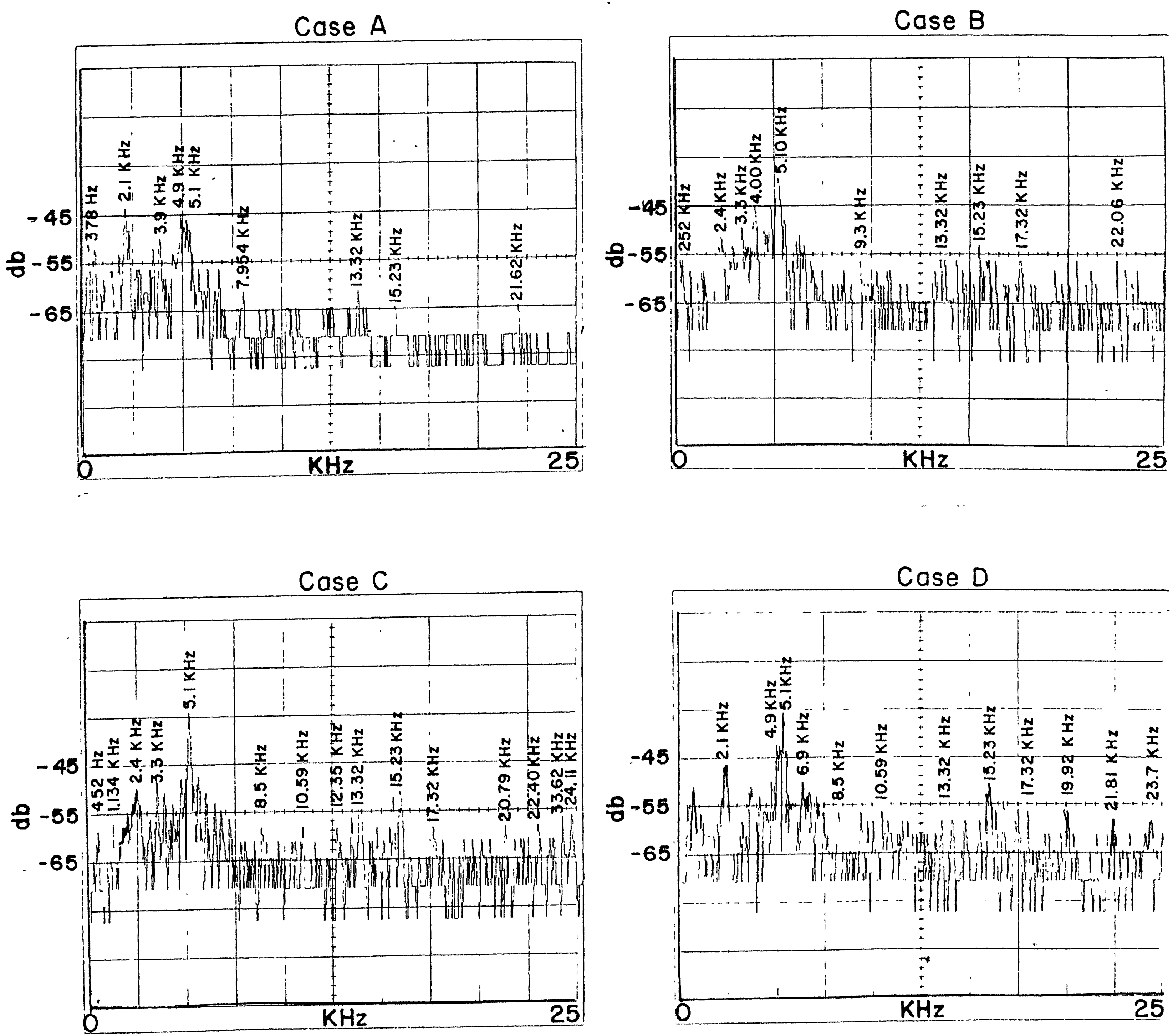

Fig. 7.17 FREQUENCY SPECTRUM AT 600 R.P.M (HIGH FREQUENCY RANGE)

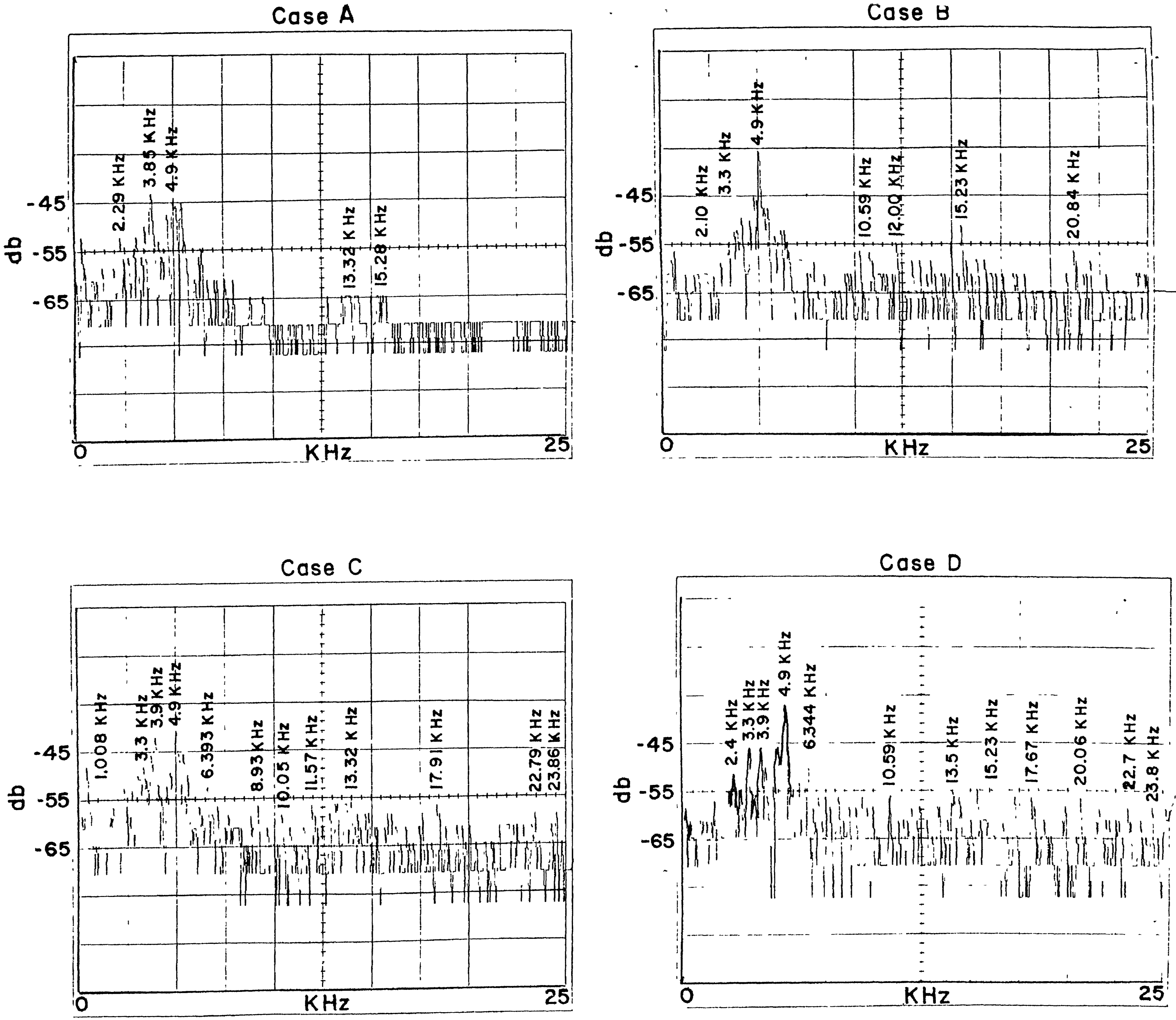

Fig 3.18 FREQUENCY SPECTRUM AT 800 R P M(HIGH FREQUENCY RANGE)

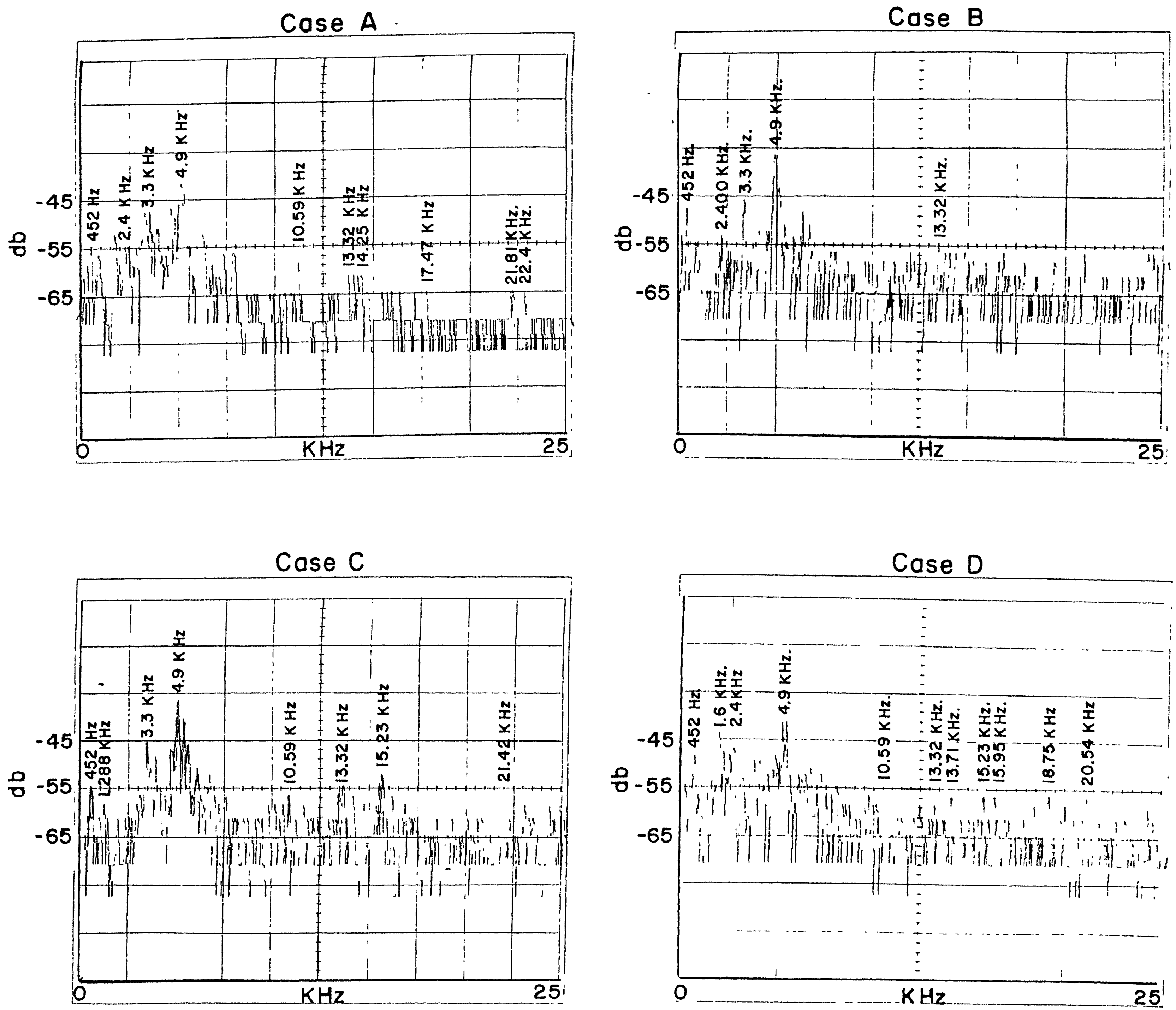

Fig.3.19 FREQUENCY SPECTRUM AT 1000 R.P.M(HIGH FREQUENCY RANGE)

rotational speeds. These frequencies are generally absent in cases B & D. These phenomenon may be due to the fact that the absence of one ball significantly alters the bearing stiffness and gives rise to distinct natural frequencies which are absent in cases B & D where the bearing stiffness is not significantly altered by these faults.

The nature of fault is more severe and complex in cases B & D so more energy is dissipated in the higher frequency zone resulting in higher peaks between 7 khz and 18 khz. Beyond 18 khz vibration in cases B & D is scattered (only one or two peaks are seen and few more at higher rotational speeds).

However in case C vibration for the same probable reason stated earlier, beyond 18 khz is considerable.

3.5 REMARKS

It can be thus observed that while the low and medium frequency range analysis does reflect the change in bearing health, the signals are however accompanied with various other characteristic

vibration features of the installation, making the fault detection difficult. The high frequency analysis clearly reflects the changes in bearing health. The severity of the fault can be assessed by the level of energy dissipation in this zone. Though the high frequency spectra may give some indication about the gravity of fault, no definite conclusion can be made about the type of fault and its location on the bearing.

CHAPTER 4

CONCLUSIONS

A bearing test rig with a geared rotor was fabricated and installed. Instrumentation was set up. Investigations were carried out for three kinds of defective bearings and their spectra were compared with that obtained from healthy bearings. Investigations were carried out at various speeds of rotor.

Characteristic frequencies were identified for observation and monitoring. These frequencies were the natural frequencies of the installation, bearing impact rates and gear tooth meshing frequencies. Analysis was carried out in the low, medium and high frequency zones. It was observed that high frequency analysis renders a better domain for monitoring bearing health in comparison to low and medium frequency zones. It is also observed that conclusions can be made regarding the gravity of bearing defects from vibration analysis. The inception of the fault and its

progress can thus be monitored keeping the healthy bearing data as reference. However, nothing can be definitely said about the location of the fault on the bearing and its type. This, is nevertheless in keeping with a practical situation where a number of faults would simultaneously be on the bearing.

4.1 LIMITATIONS

The experimental investigations were constrained by the following limitations.

a) The test rig was not ideal. A perfectly aligned rig configuration could not be achieved.

b) The rectifier designed and fabricated to convert A. C. current to D.C. did not have a R-C filter and the output current was wavy.

c)The gear tooth profile was not ideal and backlash existed even after loading the drum. d) The epoxy glue which was used to cement accelerometer with the bearing housing had limitation of giving accurate results only up to 28 khz. Frequencies beyond this could not be studied.

e) Electrical & mechanical noise was present in the system. These imperfections in the system resulted in unwanted vibration components making it more complex for analysis.

4.2 SCOPE FOR FUTURE WORK

All the spectra obtained were transferred to a computer via GPIB interface in digital form. This data can be used for computerized analysis. An expert system can be developed which processes the data monitors the trend, compares the spectrums of faulty bearing with healthy bearing,takes expert information in an interactive manner and and establishes some standard for diagnosis of defects.

The present study, was restricted to only bearing defects. The same rig can be employed in future to develop a more comprehensive diagnostic schedule to incorporate a variety of structural faults in the shafting, gearing, coupling and motor.

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APPENDIX 1

SPECIFICATIONS OF EQUIPMENT

1. DIGITAL STORAGE OSCILLOSCOPE (4072)

   MAKE - L & T Gould

   INPUT - 2 Channel BNC connectors.

   BANDWIDTH:

   DC: 0-100 Mhz(-3 db)

   AC: 4 hz-100 Mhz(-3 db)

   OPERATING TEMPERATURES: 0 Deg C to 50 deg C.

2. CHARGE AMPLIFIER TYPE 2626

   MAKE - Bruel & Kjaer, Denmark

   Charge Input - 100000 PC max

   Max Input - 10 v (10 mA) peak for less than 1 % distortion.

   Frequency Range - 0.5 hz to 100 khz

3. ACCELEROMETER

   MAKE - Bruel & Kjaer

   Charge Sensitivity - 0.119 PC/ms

   Frequency Range - 0 to 85 khz

4. TACHOMETER

   MAKE - Teclock Corporation, Japan

   Range - 0 to 10000 rpm